NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

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TESTS TO DETERMINE THE EFFECT OF HEAT ON THE PRESSURE DROP THROUGH RADIATOR TUBES

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SUMMARY

Tests to determine the effect of heat on the pressure drop through radiator tubes were made to establish the adequacy of previously developed theory. Tubes of various length-diameter ratios typical of those in current practice were tested through a wide range of heat-input rate. The tube entrance Mach number was varied from 0.12 to the value for choking at which sonic velocity was attained at the tube exits. At usual radiator operating temperatures and Mach numbers the addition of heat produced large increases in the pressure drops required to induce given flows of cooling air through the tubes. These increments in pressure drop were in good agreement with increments obtained from theoretical calculations of the heating effect. For very high tube temperatures corresponding to high rates of heat input the experimental pressure-drop increments for tubes of large length-diameter ratios exceeded the theoretical values by an appreciable amount in the vicinity of the choking Mach numbers. The experimental choking Mach numbers, however, agreed with the values predicted by theory for all tubes tested.

INTRODUCTION

A problem in the design of cooling systems for modern high-performance airplanes is the prediction of the pressure drops required across tubular radiators to induce the necessary mass flows of cooling air. The primary effect of heating a radiator tube is known to be an increase in the pressure drop required to force a given mass of cooling air through the tube.

In reference 1 a simplified theoretical method of evaluating the required pressure drop across a heated radiator is presented in the form of curves from which the pressure drop can be obtained if the radiator dimensions, the rate of heat input, the pressure and temperature ahead of the radiator, and the required rate of mass flow of air through the radiator are known. The theoretical pressure-drop characteristics given in reference 1 show the compressibility effects present at high Mach numbers as well as the heating effects and were in excellent agreement with experimental characteristics for unheated radiator tubes. No comparable experimental data were available for heated tubes. The existing data on the heating effect were generally limited to low tube Mach numbers and in many cases the heating effect could not be isolated because of the existence of other variables. (See references 2 and 3.)

The purpose of the present investigation is to determine experimentally the pressure-drop characteristics of heated radiator tubes in order to establish the adequacy of the theory of reference 1 for use in predicting the effects of heating. Tubes of several length-diameter ratios typical of those in present use for airplane radiators were tested. For each tube the mass flow was varied from a minimum value corresponding to an entrance Mach number of about 0.12 to the maximum value that could be obtained corresponding to the choked condition in which sonic velocity is reached at the tube exit. The heat input was varied from low values to values corresponding to tube temperatures appreciably higher than the temperatures at which airplane radiators are usually operated.

For ease of comparison the same symbols and radiator tube stations are used herein as were used in reference 1.

SYMBOLS

A cross-sectional area of radiator tube, square feet

a velocity of sound in air, feet per second

C_{D_f} skin-friction drag coefficient $\left(\dfrac{D_f}{q_{r_2}A_{r_2}}\right)$

c_p specific heat of air at constant pressure, Btu per pound per °F (0.24)

D_f drag force due to skin friction, pounds

d radiator-tube diameter, feet

g acceleration of gravity, feet per second per second

H heat added in radiator, Btu per second

L	length of radiator tube, feet
M	Mach number (v/a)
m	mass-flow rate, slugs per second (ρAv)
p	static pressure, pounds per square foot
Δp	total pressure at station 2 minus static pressure at station r_3, pounds per square foot
q	dynamic pressure, pounds per square foot $\left(\frac{1}{2}\rho v^2\right)$
R	Reynolds number $\left(\frac{\rho vd}{\mu} \text{ or } \frac{md}{A_{r_2}\mu}\right)$
S	area of inside surface of radiator tube, square feet
T	free-stream air temperature, $^{\circ}F$ absolute
v	velocity in radiator tube, feet per second
ρ	density, slugs per cubic foot
μ	viscosity of air, pound-seconds per square foot
γ	ratio of specific heat of air at constant pressure to specific heat of air at constant volume (1.4)

Subscripts

i	low speed, condition for incompressible unheated flow
2	station ahead of radiator
r_2	within radiator at tube entrances
r_3	within radiator at tube exits

TEST APPARATUS

A sketch of the test setup is shown as figure 1. Air from the compressor flowed into the supply tank where any condensed moisture settled out. The air then passed through a steam radiator and two pressure-regulator valves into a surge tank, the volume of which was approximately 3.5 cubic feet. A 100-mesh screen was installed in the surge tank. Air from the surge tank entered the radiator tube through a bell-mouth entrance and exhausted out of the rear of the tube to the atmosphere. The rate of air flow through the radiator tube was controlled by varying the pressure in the surge tank. The air in the surge tank was maintained at the desired pressure by use of the two pressure-regulator valves. The temperature of the air could be raised by the steam radiator to a value high enough to prevent condensation of moisture in the expansion occurring at the tube entrance.

Tubes of approximately 6-, 12-, 18-, and 24-inch lengths having length-diameter ratios of 29.25, 58.50, 87.75, and 117.00, respectively, were tested. The tubes were made from 0.250-inch Inconel tubing which was reamed and polished to a constant inside diameter of 0.205 inch. Inconel was used because of its high electrical resistivity since the tubes were heated by passing an electric current through them. This method of heating gives an approximately constant rate of heat input per unit length of tube. The electrical circuit is shown in figure 1. The power was supplied by a bank of storage batteries, and the flow of current was regulated by means of a slide-wire rheostat. The power input was measured with an ammeter in series with the tube and a voltmeter which measured the voltage drop across the tube.

Thermocouples were silver soldered on the tubes at the approximate locations shown in figure 1. Stagnation temperature ahead of the tube was measured by means of a thermocouple installed in the surge tank. Static-pressure tubes were installed in the surge tank and bell-mouth throat (fig. 2).

The radiator tube was insulated thermally from the surge tank with an asbestos gasket. (See fig. 2.) A plywood shield was installed completely around the tube, as shown in figure 1, to provide a dead-air space and to prevent drafts from affecting the temperature of the tube. A thermocouple installed on the shield was used in conjunction with the three tube thermocouples to measure the heat losses during the tests.

TEST METHOD

Each of the bell-mouth entrances to the radiator tubes was calibrated for the unheated condition so that the actual mass flow could be determined from the pressure and temperature measurements at the tube entrance in the surge tank. The calibration was made in the following manner: The supply tank was filled with air to a pressure of approximately 150 pounds per square inch and the pressure and temperature of the air in the supply tank were recorded. Air was then allowed to flow through the supply line, into the surge tank which was held at constant pressure by the regulator valves, and through the radiator tube into the atmosphere. While air was flowing through the tube, the pressures at stations 2 and r_2 and the temperature at station 2 were recorded. After the air pressure in the supply tank dropped to approximately 50 pounds per square inch, the flow of air was stopped. The total time that the air had been flowing was measured and recorded and the pressure and temperature of the air in the supply tank were again recorded. Since the volume of the tank was known, the amount of air that flowed out of the supply tank could be computed. This procedure was repeated for various pressure drops across the tube over its test range. The mass flow was computed from the pressures measured at stations 2 and r_2, the temperature at station 2, and the area of the tube. The calibration constant, or ratio of actual mass flow to calculated mass flow, was found to vary between 0.97 and 1.00 for the various entrances.

A calibration of the heat losses for various operating tube temperatures was made as follows: Various amounts of power were put into the tube and the stabilized tube and shield temperatures were measured with no air flowing through the tube. The average tube temperature minus the shield temperature was then used as an index of the heat loss. During the tests it was assumed that with air flowing through the radiator tube, a given average tube temperature minus shield temperature indicated the same loss as that obtained with the no-flow condition. These losses were subtracted from the total heat input to the tube to get the actual heat input into the air. Although air flowing through the tube changed the temperature distribution along the tube, the error occurring in the use of the average tube temperature to determine the heat losses was small and therefore was neglected. The losses for the tubes tested varied from approximately 2 percent to approximately 25 percent of the total power input depending upon the length of the tube, the entrance Mach number, and the rate of heat input. The highest percentage loss occurred for the shortest tube at low Mach numbers. With increasing Mach number and increasing tube length the heat loss decreased rapidly, and in all cases the loss was relatively small near choking conditions.

Preliminary calculations showed that the air would reach the saturation condition in the tube entrances at the higher Mach numbers. In order to prevent this condition it was decided to heat the entering air. A small tubular radiator was designed and constructed for this purpose and steam was used as the heating medium. Preliminary check tests were made with and without heating to determine the effect of condensation. Test conditions were chosen for which saturation temperatures would be encountered if the air were not heated. The results indicated that the tube pressure drop for a given Mach number was the same regardless of whether preheating was used. A similar result was obtained in the investigation in reference 4. The data shown in the present paper were obtained without preheating the air. Inlet-air temperatures averaged about 80° F.

RESULTS AND DISCUSSION

Evaluation of Δp_i.- The effects of heating and compressibility on the pressure drop through tubes can be conveniently expressed in terms of the ratio of the actual pressure drop for a given operating condition Δp to the "incompressible" pressure drop Δp_i which would exist if no heating or compressibility effects were present. (See reference 1.) The value of Δp_i is easily computed if L/d and the Reynolds number for the tube are known. Equation (2a) of reference 1 gives the following expression for Δp_i

$$\Delta p_i = \frac{1}{2\rho_2}\left(\frac{m}{A_{r_2}}\right)^2\left(1 + C_{D_{f_i}}\right) \qquad (1)$$

In equation (1) the drag coefficient of the tube $C_{D_{f_i}}$ depends only on L/d and the known Reynolds number and can be obtained directly from figure 2 of reference 1. The mass flow m and the density ρ_2 are known for the operating condition under consideration. After Δp_i is calculated by this method, the actual pressure drop Δp can be determined if the ratio $\Delta p/\Delta p_i$ is known. Reference 1 shows that $\Delta p/\Delta p_i$ for a given tube is a function of only the entrance Mach number M_{r_2} and the heat-input factor $\frac{H}{c_p g m T_{r_2}}$. The results of the present tests are therefore given as plots of $\Delta p/\Delta p_i$ against Mach number M_{r_2} for various values of $\frac{H}{c_p g m T_{r_2}}$.

The computation of Δp_i was essential in the reduction of the data to the form $\Delta p/\Delta p_i$. The value of Δp_i could have been computed from the equation just given by use of values of $C_{D_{f_i}}$ obtained from figure 2 of reference 1. It was considered more accurate, however, to base the calculation on values of $C_{D_{f_1}}$ actually measured for the tubes tested. Tests without heat were therefore made for the same range of test Mach number for which tests were later made with heat. The value of $C_{D_{f_1}}$ was obtained from measurements of the mass flow and pressure drop by means of the following relation from reference 1:

$$C_{D_{f_1}} = \frac{1}{7}\left(1 + \frac{5}{{M_{r_2}}^2}\right)\left[1 - \left(\frac{\rho_{r_3}}{\rho_{r_2}}\right)^2\right] + \frac{6}{7} \log_e\left(\frac{\rho_{r_3}}{\rho_{r_2}}\right) \tag{2}$$

The entrance Mach number in equation (2) was obtained from the known mass flow, temperature, and pressure p_{r_2}, as shown in reference 1. The density ratio ρ_{r_3}/ρ_{r_2} needed for solution of this equation was obtained from simultaneous solution of the energy equation for an unheated tube

$$\frac{{v_{r_2}}^2}{2} + \frac{\gamma}{\gamma - 1}\frac{p_{r_2}}{\rho_{r_2}} = \frac{{v_{r_3}}^2}{2} + \frac{\gamma}{\gamma - 1}\frac{p_{r_3}}{\rho_{r_3}} \tag{3}$$

the continuity relation

$$\rho_{r_2} v_{r_2} = \rho_{r_3} v_{r_3}$$

and the expression for the velocity of sound

$$a_{r_2} = \sqrt{\gamma\frac{p_{r_2}}{\rho_{r_2}}}$$

For $\gamma = 1.4$, the solution for ρ_{r_3}/ρ_{r_2} is

$$\frac{\rho_{r_3}}{\rho_{r_2}} = \frac{2M_{r_2}^{\ 2}}{-5\left(p_{r_3}/p_{r_2}\right)+\sqrt{25\left(p_{r_3}/p_{r_2}\right)^2 + 4M_{r_2}^{\ 2}\left(M_{r_2}^{\ 2} + 5\right)}} \qquad (4)$$

The quantity M_{r_2} in equation (4) was obtained from the measured entrance conditions; p_{r_2} and p_{r_3} were measured directly.

Tubes with smooth entrances.- Values of $C_{D_{f_1}}$ were first established for tubes of various length which had round smooth entrances (fig. 3). Extensive laminar flow existed. This condition is most apparent for the shorter tubes in which a larger fraction of the tube length was subjected to laminar flow. At Reynolds numbers above approximately 40,000 and 22,000 for the tubes for which $\frac{L}{d} = 29.25$ and $\frac{L}{d} = 58.50$, respectively, the point of transition from laminar to turbulent flow moved toward the tube entrance as the Reynolds number was further increased. At the highest test Reynolds number (approximately 100,000) the experimental curves approach the curves for completely turbulent flow determined from the equation

$$C_{D_{f_1}} = 0.0785\,\frac{4L}{d}\,R^{-0.25} \qquad (5)$$

The entrance Mach number was almost the same at a given Reynolds number for both the shorter tubes for Reynolds numbers up to about 60,000 at which choking begins in the tube for which $\frac{L}{d} = 58.50$. Thus, the critical Reynolds numbers shown are believed to be not influenced by Mach number.

A comparison between experimental and theoretical static-pressure-drop ratios for given heat-input factors is presented in figure 4 for the smooth-entrance tube for which $\frac{L}{d} = 29.25$ with laminar flow present. At zero heat input the theoretical results were expected to agree with the experimental results because the drag coefficient $C_{D_{f_i}}$ was computed from data obtained for the unheated condition as previously discussed. When heat is added to the tube, however, large deviations occur between the theoretical and experimental results. For the heated condition fluctuating data were obtained which could not be duplicated in repeat tests. The addition of heat at a fixed entrance Mach number and Reynolds number apparently caused the point of transition from laminar to turbulent flow to shift toward the tube entrance, and thus to increase the friction and heat-transfer coefficients. This result was indicated not only by the fact that the pressure-drop ratios (fig. 4) are higher than those predicted by the theory (which assumes no change in skin-friction coefficient) but also by fluctuations in the tube temperature for a given heat input, lower temperatures occurring for the higher pressure drops. Since the assumption was made in reference 1 that addition of heat has no effect on the friction coefficient, the results of tests in which the friction coefficient is changed by the addition of heat should not be expected to agree with the theory.

<u>Tubes with transition fixed</u>.- In an actual airplane radiator the tube entrances are usually sharp edge and are rough enough to cause turbulent flow to exist for the entire length of the radiator tubes. In order to simulate this actual operating condition and to avoid the difficulties arising from the action of heating on the laminar flow the point of transition was fixed in the entrance of the tubes tested by a ring of commercial iron cement (Smooth-On No. 1) a few thousandths of an inch thick and approximately 1/32 inch wide. (See fig. 2.) The rest of the data and figures given herein are for this fixed-transition condition.

The relationship between $C_{D_{f_i}}$ and Reynolds number was established for the tubes with the transition fixed in the entrance in the same manner as that described for the tubes with smooth entrance. Slightly higher drag coefficients were obtained (fig. 5) than were predicted by the empirical relation for purely turbulent flow. The data in figure 5 were used to compute the values of Δp_i used in reducing the test data to the form $\Delta p/\Delta p_i$.

In figure 6 the variation of $\Delta p/\Delta p_i$ with heat-input factor is presented for various entrance Mach numbers for the shortest tube $\left(\frac{L}{d} = 29.25\right)$. The effect of the addition of heat is very great and becomes particularly pronounced at the higher entrance Mach numbers. The critical pressure-drop-ratio line (sonic velocity attained at the tube exit) shows that the choking entrance Mach number is considerably reduced by the addition of heat.

A comparison between experimental and theoretical static pressure-drop ratios is presented for various heat-input factors and length-diameter ratios in figure 7. For the short tube $\left(\frac{L}{d} = 29.25\right)$ good agreement is noted for all values of heat-input factor obtained (figs. 7(a) to 7(e)). With increasing length-diameter ratio and increasing heat-input factors the discrepancy between the theoretical and experimental values becomes larger The discrepancy also becomes more pronounced as the entrance Mach numbers are increased toward their choking values. The choking Mach numbers indicated in the tests were in good agreement with the theoretically predicted choking Mach numbers.

The discrepancy between theory and experiment at conditions of high heat-input rate for Mach numbers near choking is believed to be due to a simplifying approximation made in the theory. This approximation involved the assumption that the theoretical relation between the friction factor $C_{D_f}/C_{D_{f_1}}$ and density ratio ρ_{r_2}/ρ_{r_3}, a relation which was known exactly only for the unheated condition, could be used also for the heated condition. As pointed out in reference 1, this assumption will result in negligible error except for extremely high heat-input rates or for Mach numbers near choking. The magnitude of the error involved was evaluated for a flow condition at high heat input and high Mach number by computing the pressure drop by means of a step-by-step integration process along the tube, which eliminated the necessity for making the aforementioned approximation. The result obtained was in close agreement with experimental results. The discrepancies noted in figure 7 for the extreme flow conditions are thus attributable to the assumption made regarding the friction factor in the theory. As a corollary, the other important assumptions made in the theory appear to be justified. Of particular interest is the assumption that the velocity profile at any given station in the tube is not altered by the addition of heat. The results just described indicate that this assumption is correct when the tube flow is entirely turbulent. When appreciable laminar flow was present, however, the addition of heat had a marked effect on the flow stability and velocity profile, which tended to move the point of transition from laminar to turbulent flow forward towards the tube entrance, as previously described.

In the present tests heat was added to the tubes at a constant rate per unit length of tube. In an actual radiator installation heat is not added in this manner but more nearly at a constant tube-wall temperature. An investigation to determine the difference in $\Delta p/\Delta p_1$ obtained from the two methods of adding heat disclosed that for the worst condition for which data are presented (the tube for which $\frac{L}{d} = 29.25$ and heat-input factor = 0.3), the difference in $\Delta p/\Delta p_i$ was less than 2 percent of the value obtained for uniform addition of heat.

At entrance Mach numbers below the value at which choking begins, the difference between the theoretical and experimental results is small even for high heat inputs and long tubes. An example is given in reference 1 for an airplane radiator in current use. For this example, at sea level (maximum $M_{r_2} = 0.198$ and heat-input factor = 0.074) the difference between the theoretical and experimental results is less than 2 percent of the theoretical prediction of $\Delta p/\Delta p_i$. At an altitude of 30,000 feet (maximum $M_{r_2} = 0.165$ and heat-input factor = 0.314) the difference is less than 3 percent of the value of $\Delta p/\Delta p_i$ predicted by theory. It thus appears that for typical present-day practical applications the theoretical curves of reference 1 may be used directly to obtain the effects of heat and compressibility on the pressure drop in tubular radiators.

CONCLUDING REMARKS

Tests to determine the effect of heat on pressure drop through radiator tubes were made to establish the adequacy of previously developed theory. The primary effect of heating in radiator tubes is to cause a large increase in the pressure drop required to produce a given cooling-air flow. The choking or limiting entrance Mach number is reduced as heat is added. The capacity of the simplified theory to predict these effects with satisfactory accuracy for tube dimensions and heat-input rates encountered in present-day practice is established by the present test results. For heat-input rates beyond current practice the experimental pressure requirements exceeded the theoretical predictions by an amount which increased with an increase in the tube length-diameter ratio, the heat-input rate, and the entrance Mach number. The experimental choking Mach numbers, however, were in good agreement with those predicted by the theory for even the highest heat-input rates.

The fact that heating causes changes of the velocity profile was indicated at the lower Reynolds numbers when an appreciable length of laminar flow existed in the forward end of the tubes with smooth rounded entrances. When the flow through the entire length of the tubes was turbulent, the results indicated that the velocity profile was not altered by the addition of heat.

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National Advisory Committee for Aeronautics
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REFERENCES

1. Becker, John V., and Baals, Donald D.: Simple Curves for Determining the Effects of Compressibility on Pressure Drop through Radiators. NACA ACR No. L4I23, 1944.

2. Lelchuk, V. L.: Heat Transfer and Hydraulic Flow Resistance for Streams of High Velocity. NACA TM No. 1054, 1943.

3. McAdams, William H.: Heat Transmission. Second ed., McGraw-Hill Book Co., Inc., 1942, ch. VII, pp. 154-209.

4. Keenan, Joseph H., and Neumann, Ernest P.: Measurements of Friction Coefficients in a Pipe for Subsonic and Supersonic Flow of Air. NACA ARR No. 3G13, 1943.

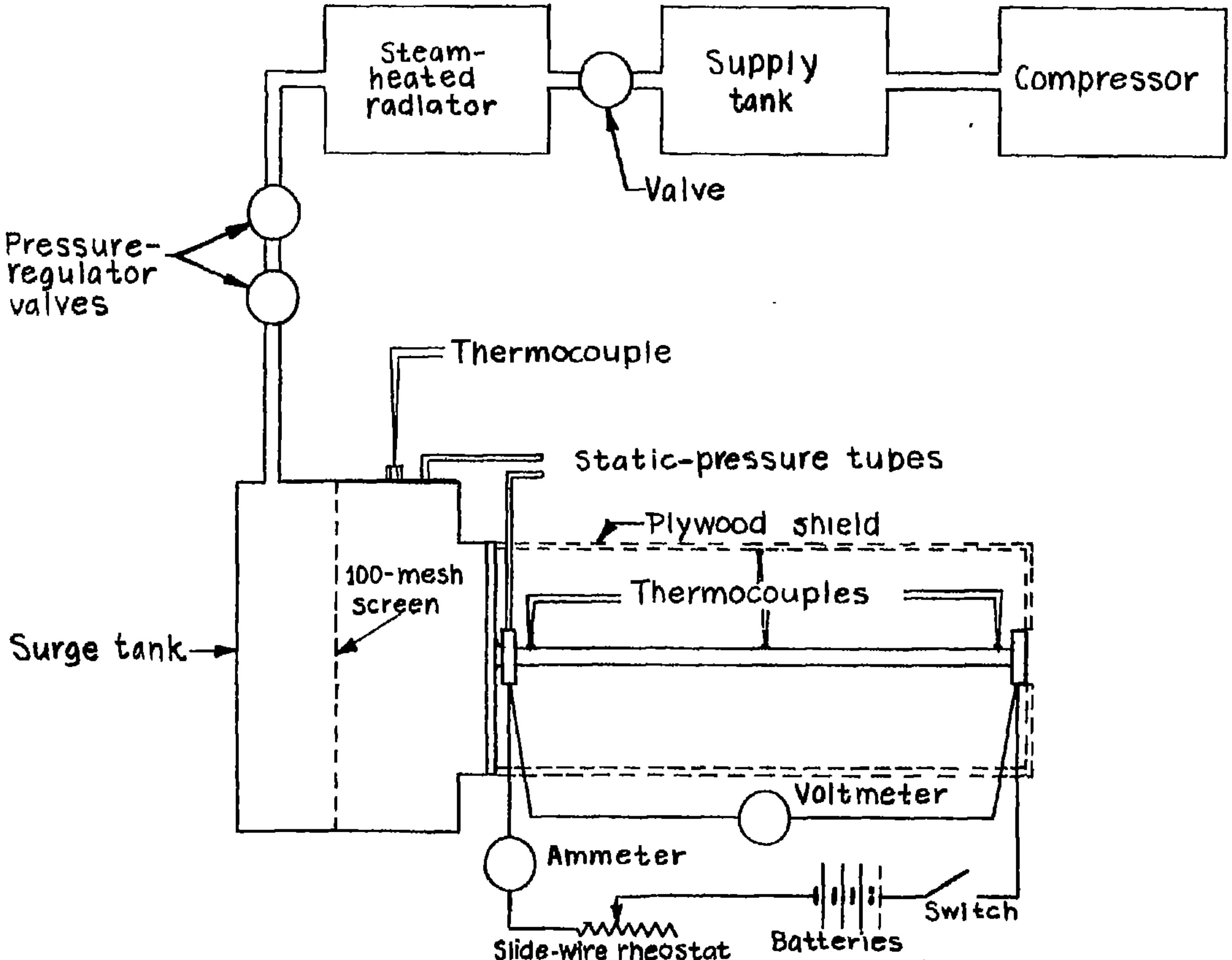

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Figure 1.- Schematic diagram of test setup.

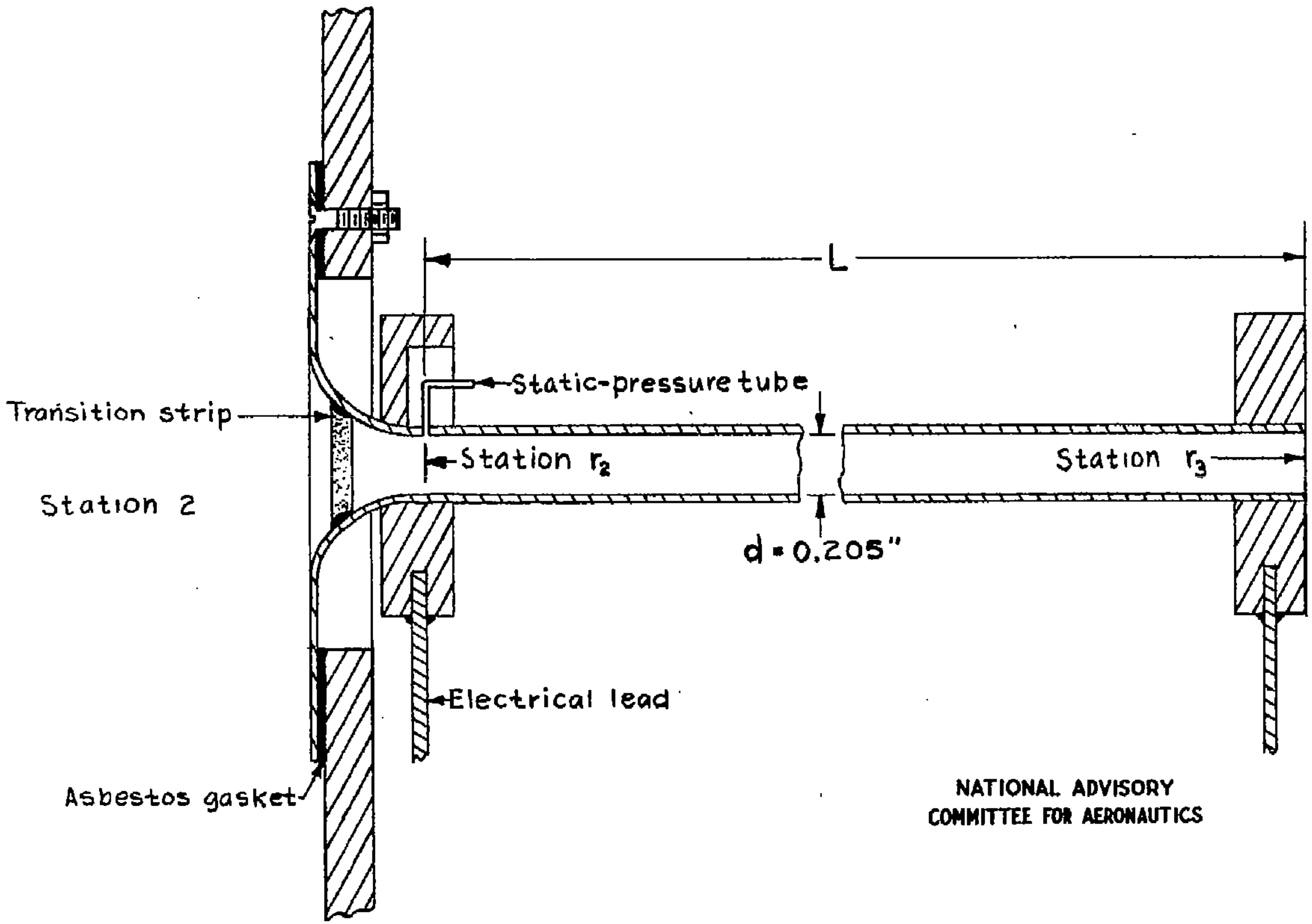

Figure 2.- Section through radiator tube.

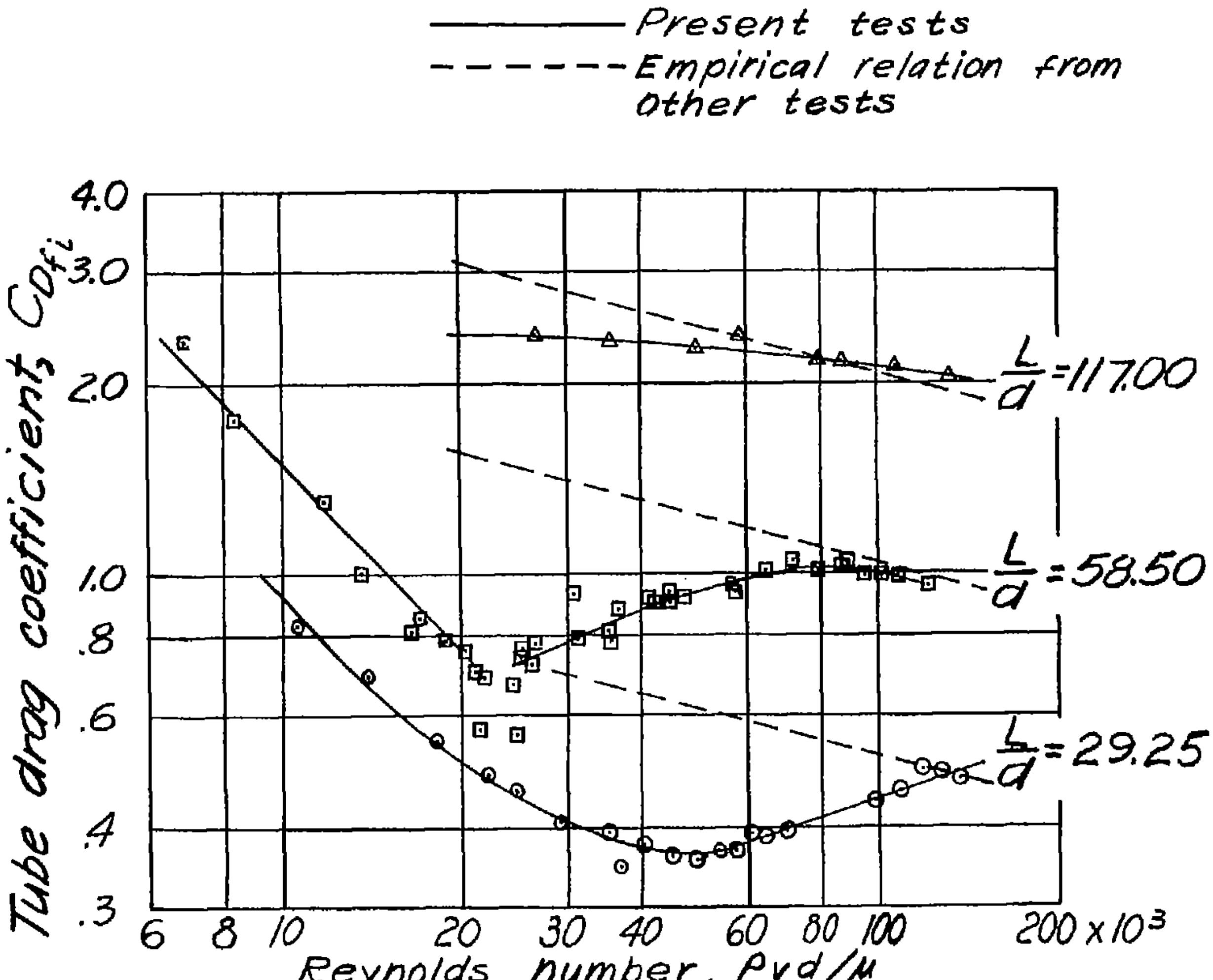

Figure 3.- Variation of tube drag coefficient with Reynolds number. Round smooth entrances; empirical curve obtained from $C_{D_{f_i}} = 0.0785 \frac{4L}{d} R^{-0.25}$

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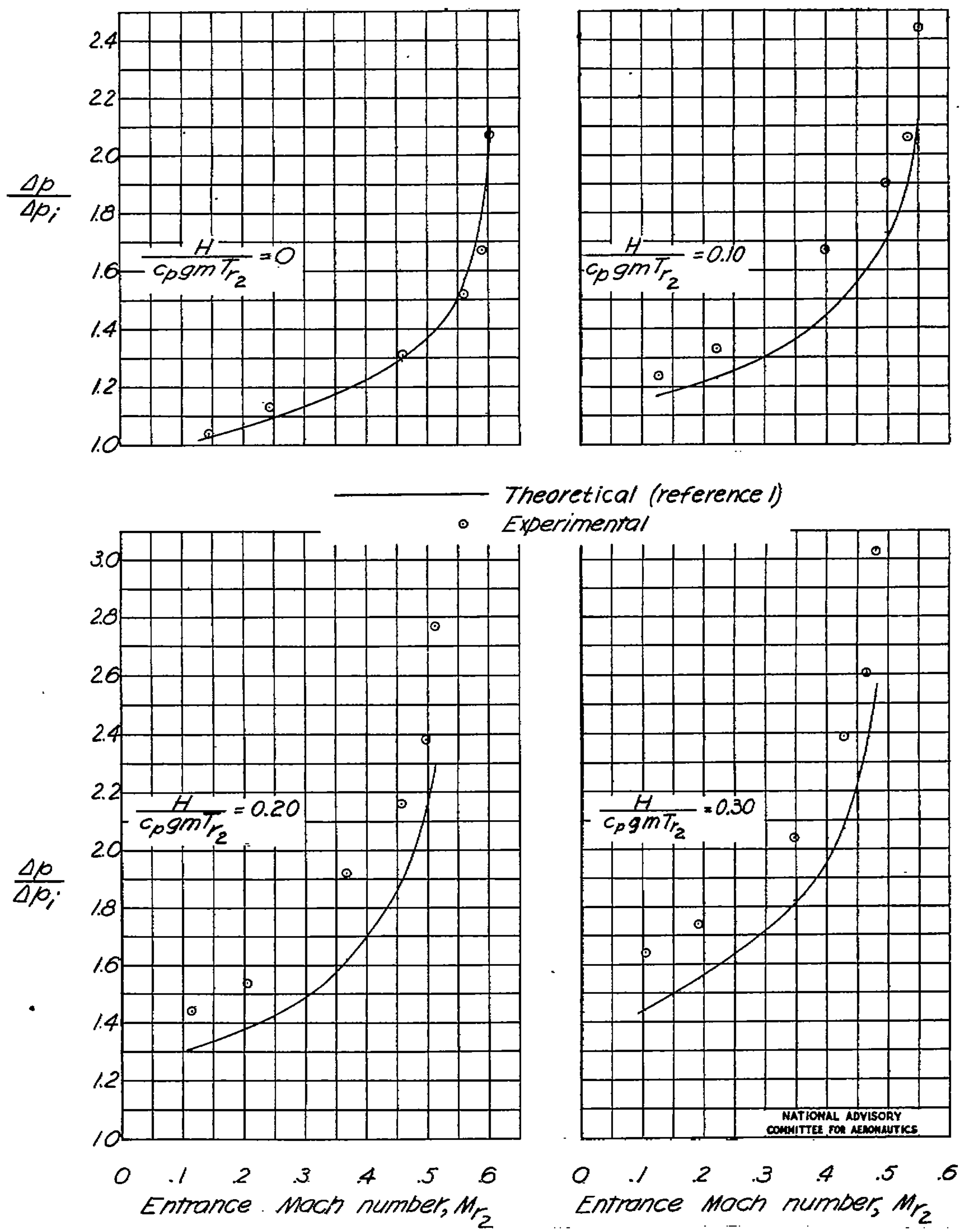

Figure 4. — Variation of static-pressure-drop ratio with Mach number at various heat-input factors. $\frac{L}{d} = 29.25$; round smooth entrance.

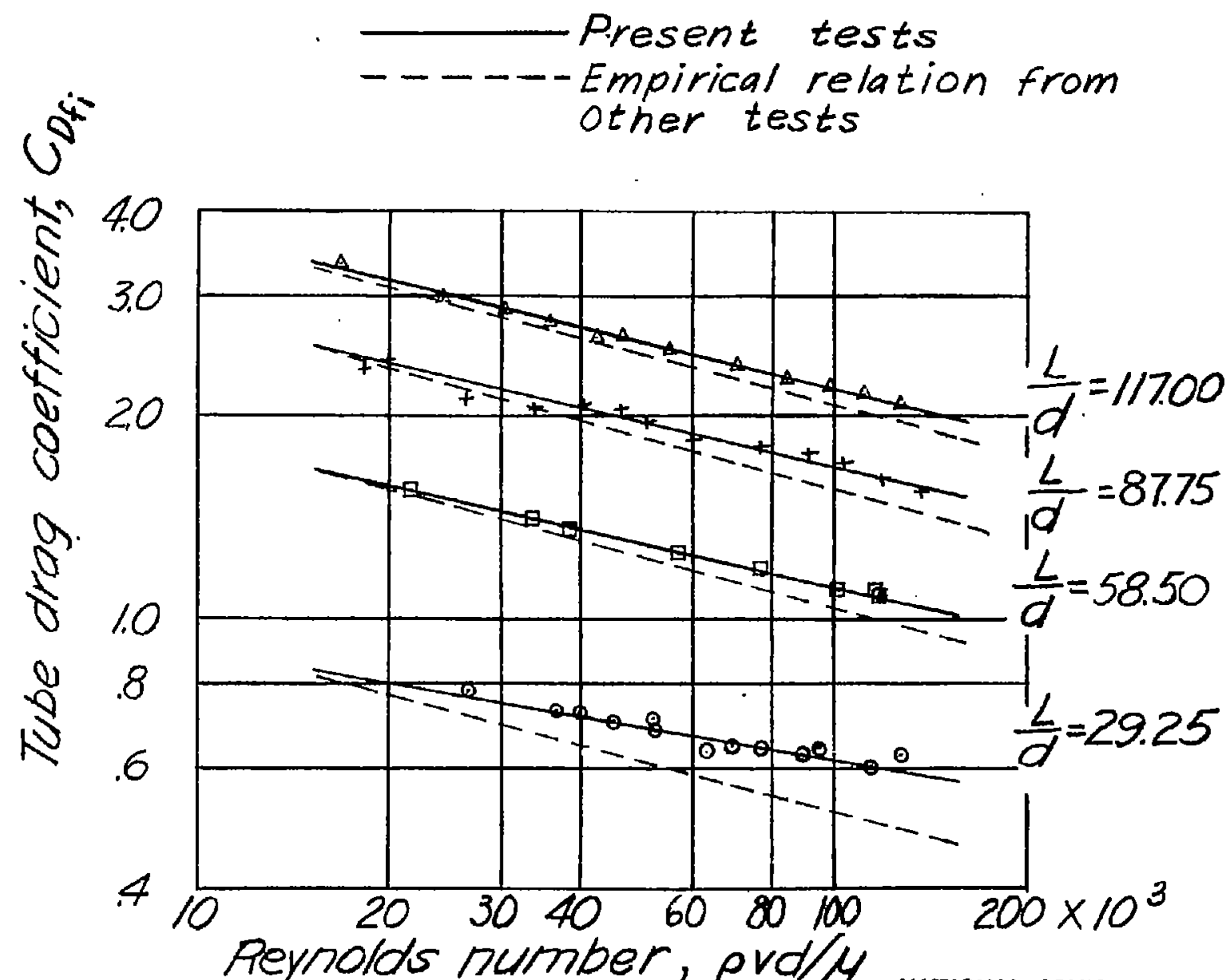

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Figure 5.—Variation of tube drag coefficient with Reynolds number. Transition fixed at the entrance; empirical curve obtained from $C_{D_{f_i}} = 0.0785 \frac{4L}{d} R^{-0.25}$.

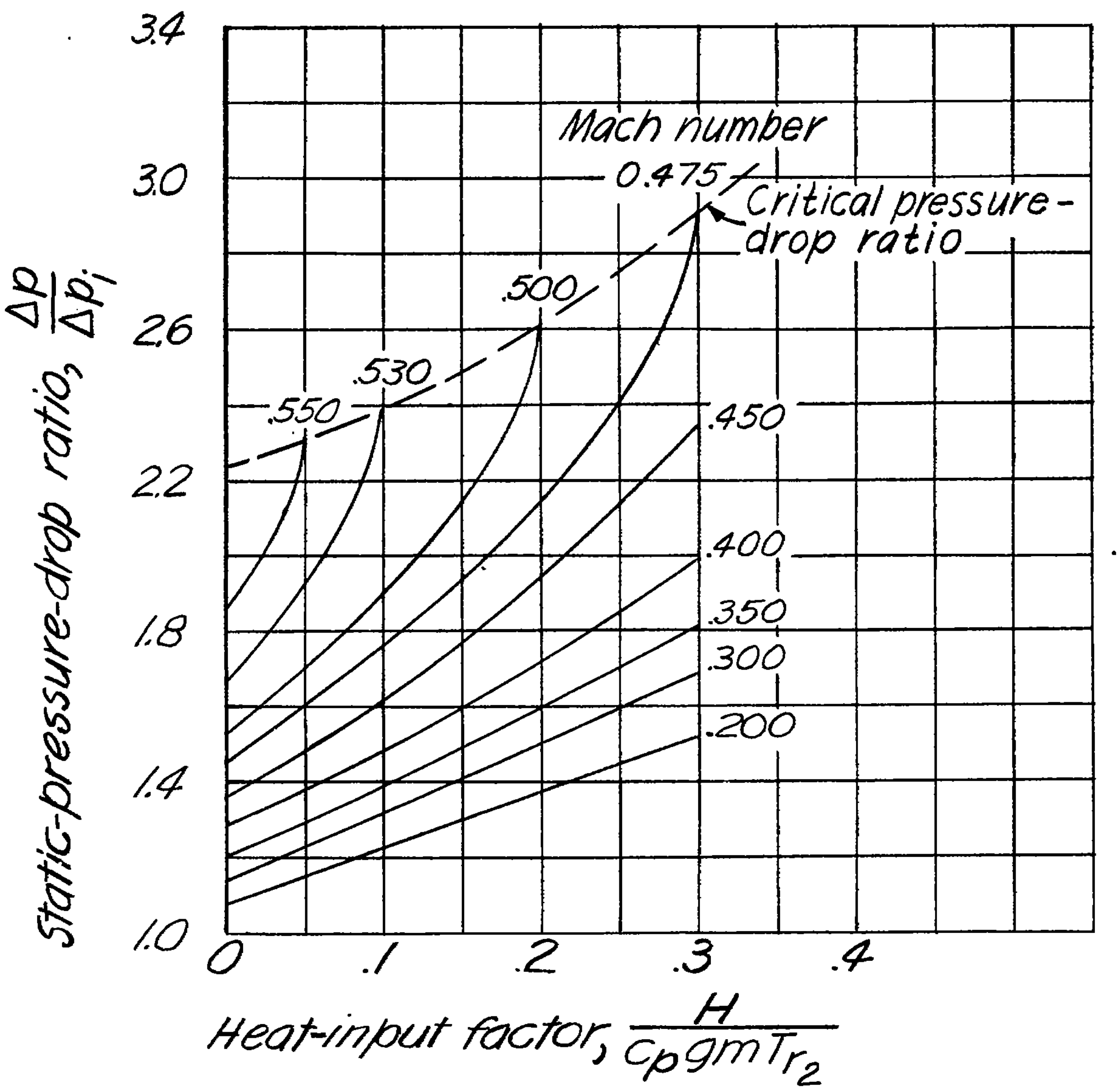

Figure 6.-Variation of static-pressure-drop ratio with heat-input factor. $\frac{L}{d} = 29.25$; transition fixed at tube entrance.

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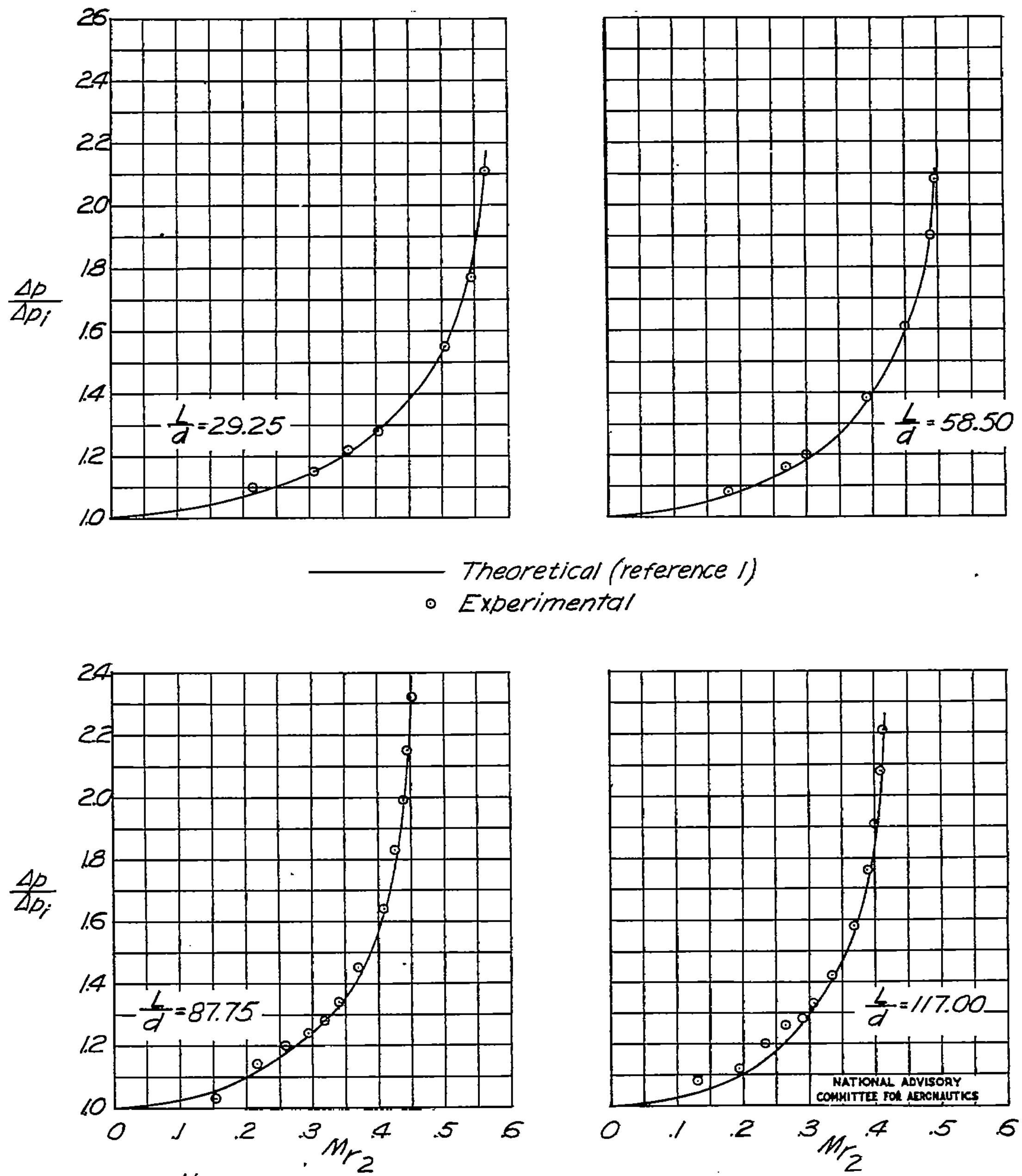
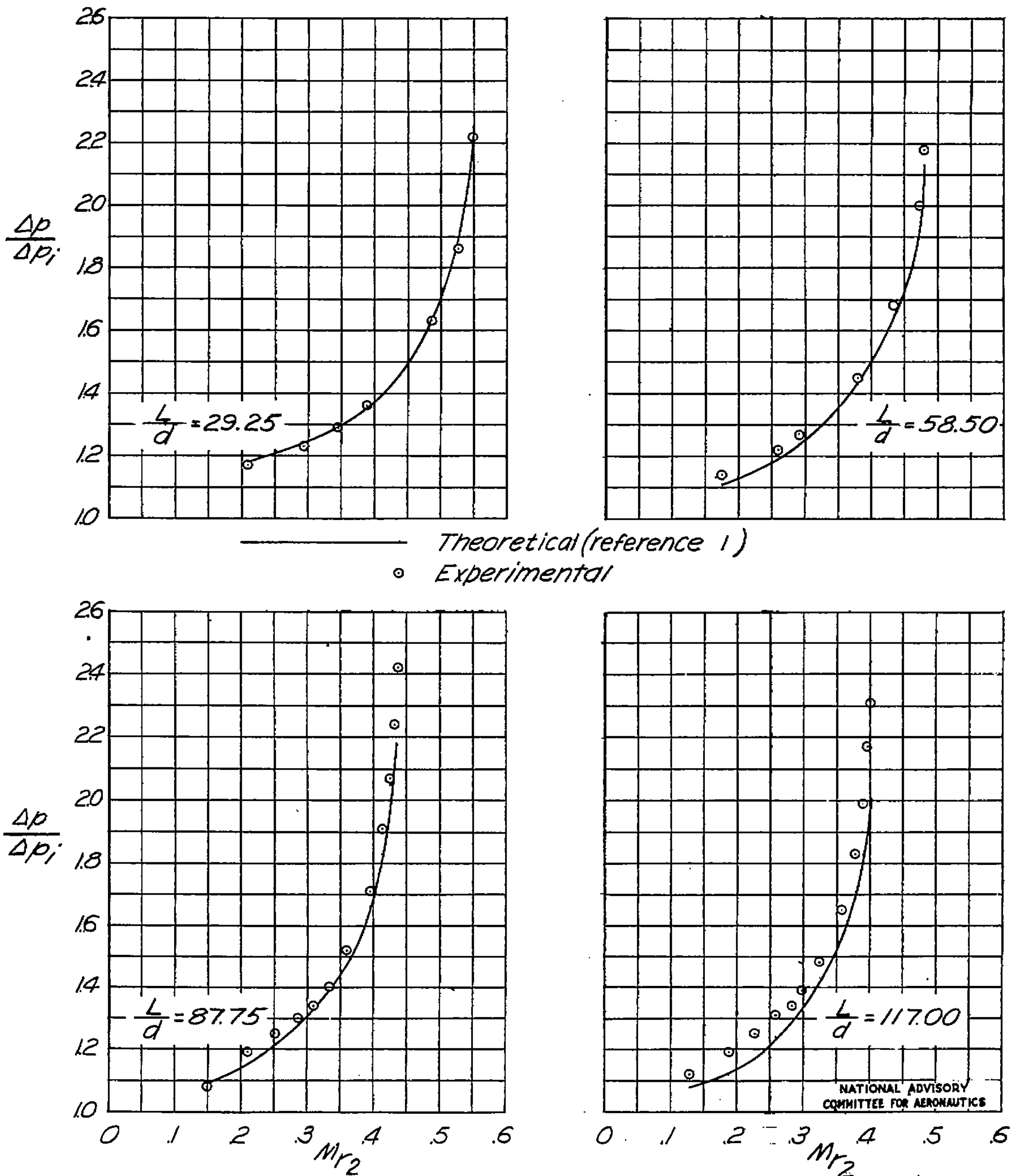


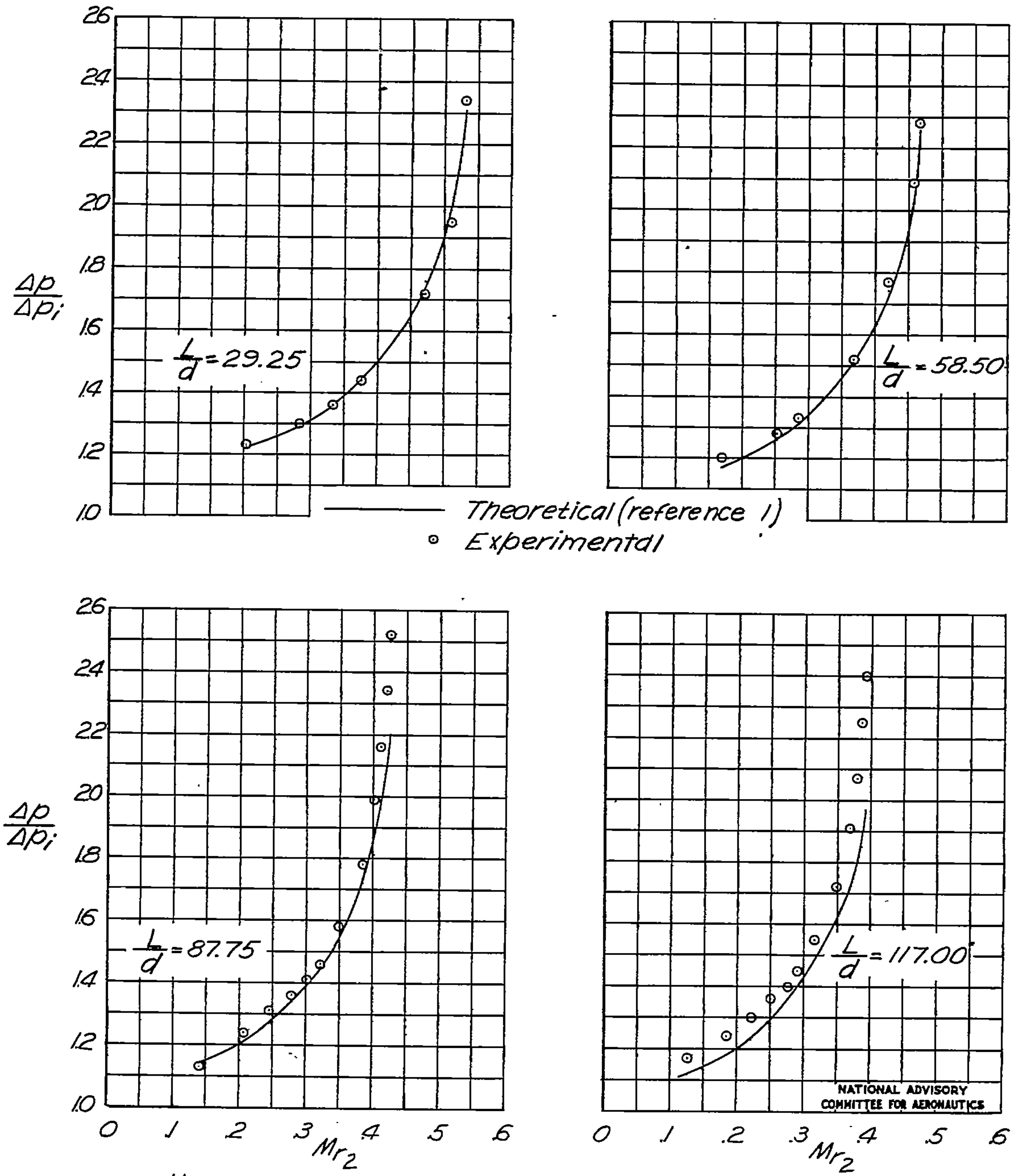
(a) $\frac{H}{c_p g m T_{r_2}} = 0$.

Figure 7. - Variation of theoretical and experimental static-pressure-drop ratio with Mach number.



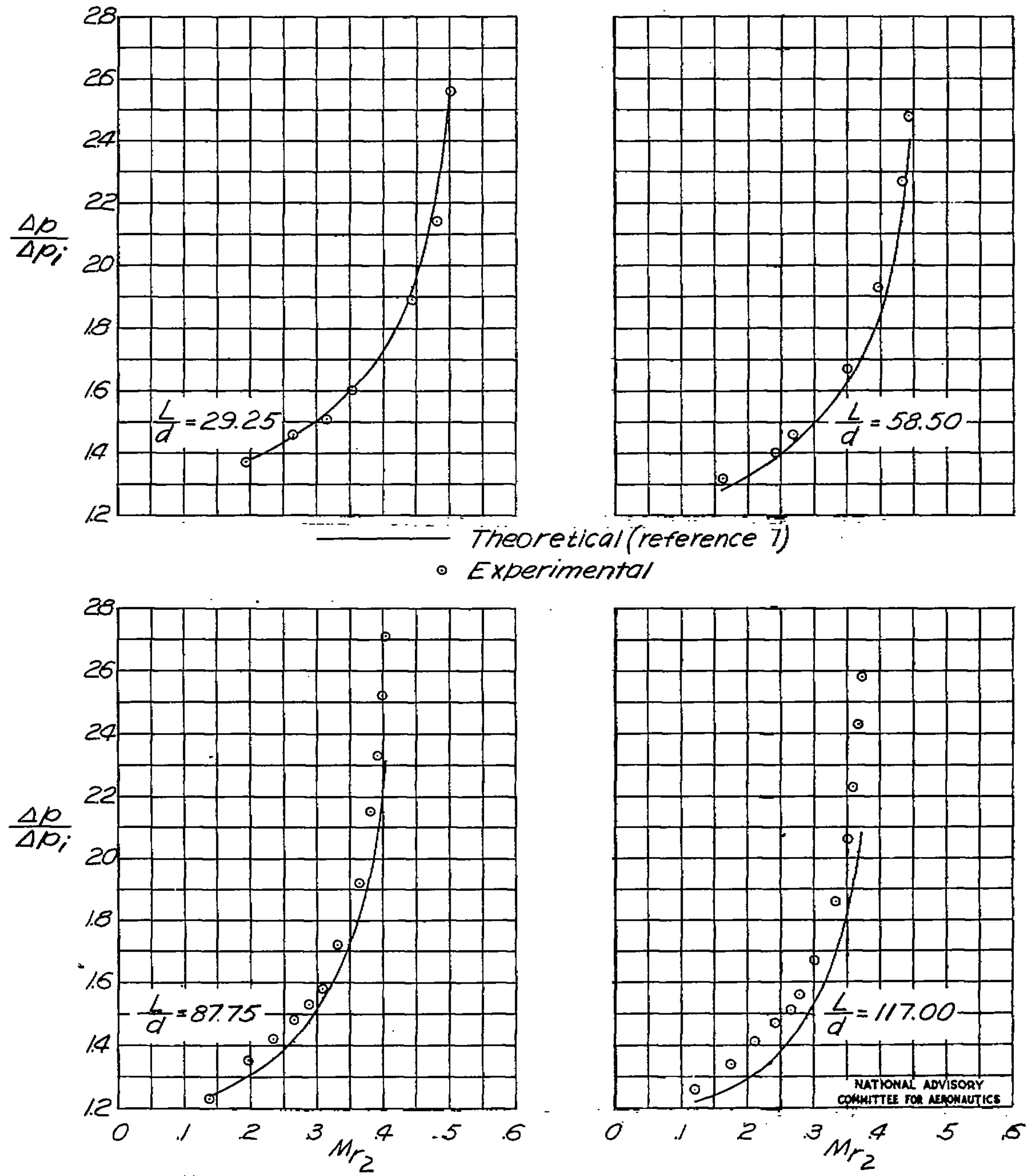
(b) $\frac{H}{c_p g m T_{r_2}} = 0.05.$

Figure 7 .— Continued.



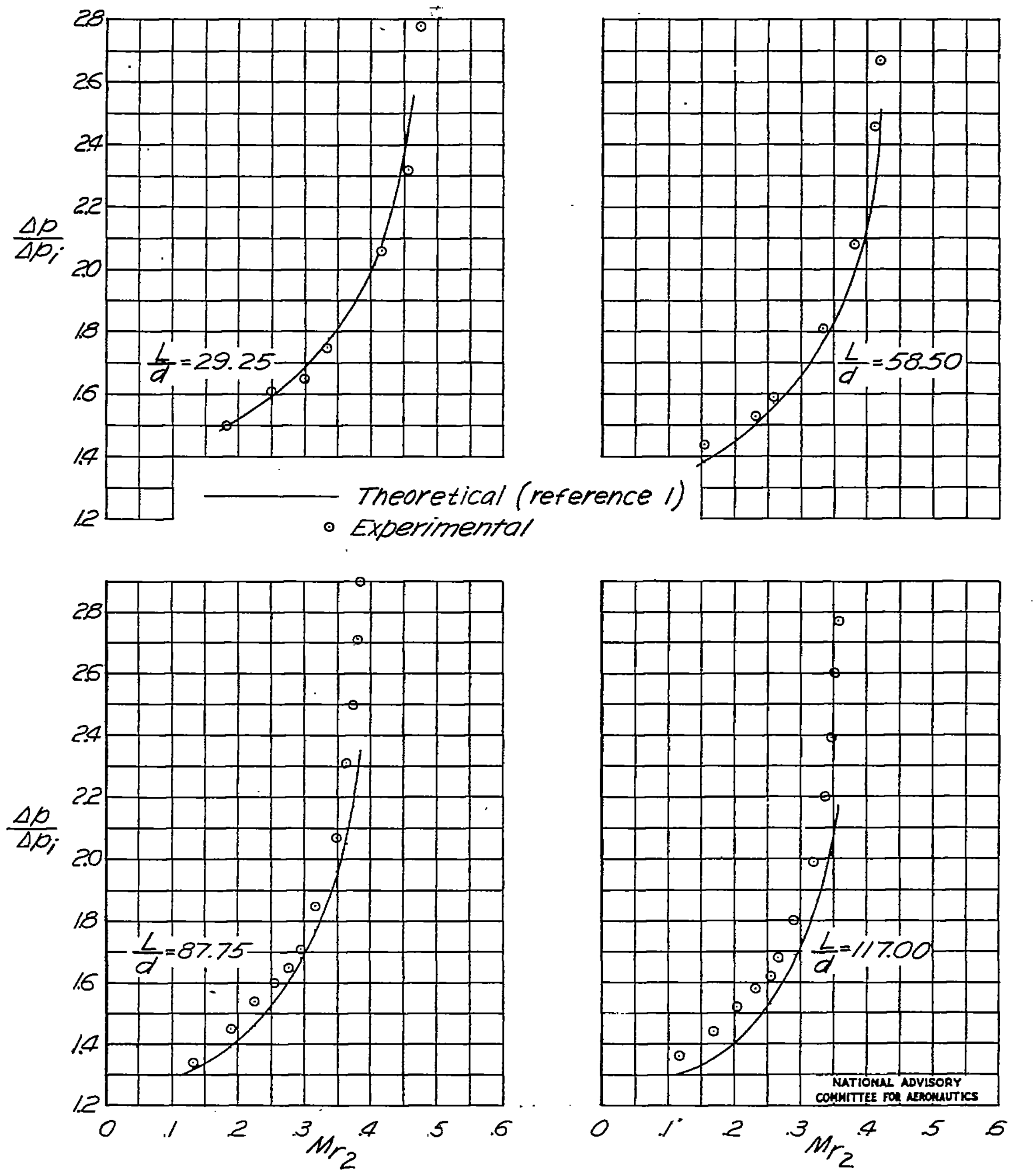
(c) $\frac{H}{c_p g m T_{r_2}} = 0.10.$

Figure 7 .— Continued.



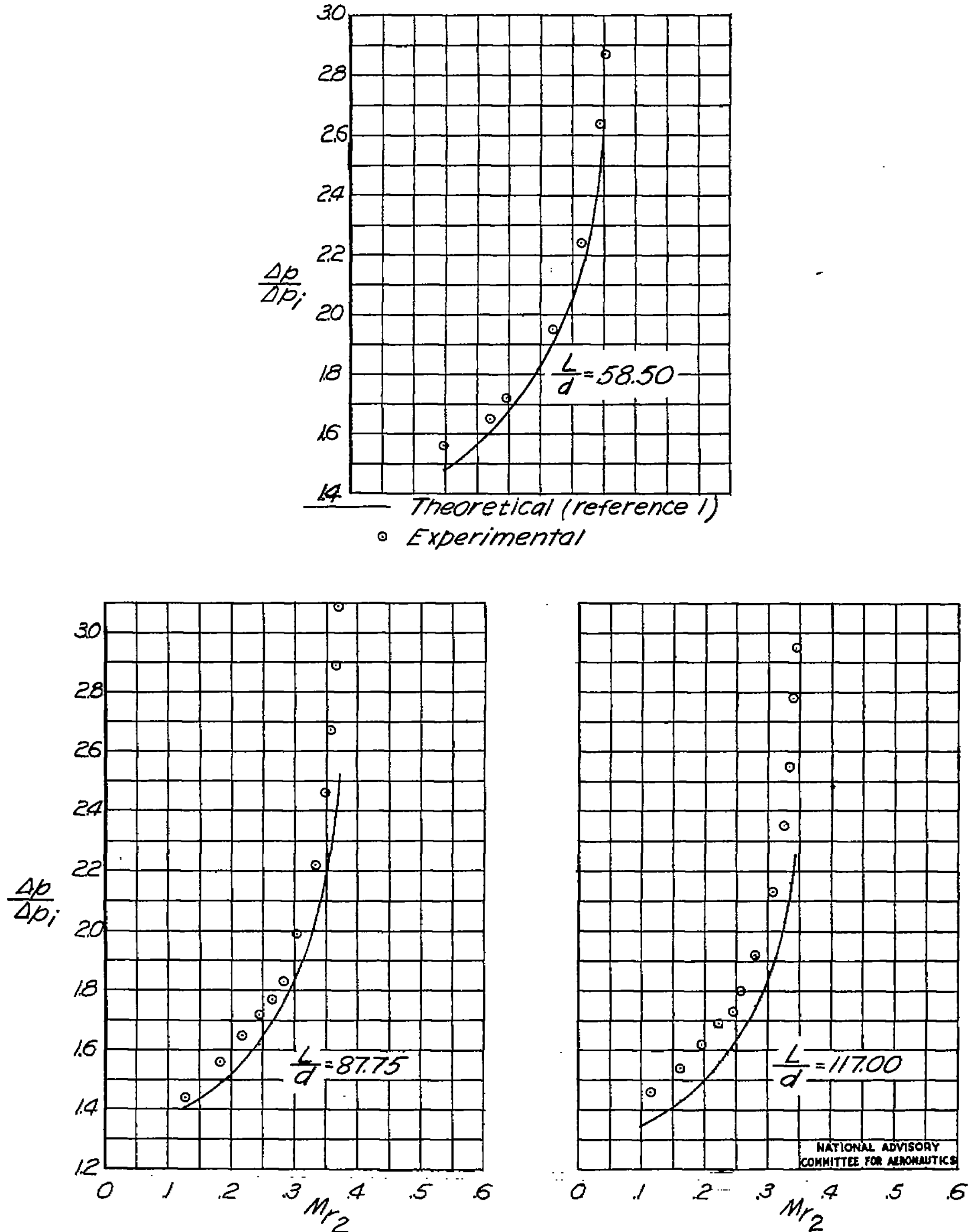
(d) $\frac{H}{c_p g m T_{r_2}} = 0.20.$

Figure 7 .— Continued.



(e) $\frac{H}{c_p g m T_{r_2}} = 0.30.$

Figure 7 .– Continued.



(f) $\frac{H}{c_p g m T_{r_2}} = 0.40.$

Figure 7.—Concluded.